Technical

Report

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A REVIEW OF RESEARCH IN THE FIELD OF GAS-LUBRICATED BEARINGS

By

Dudley D. Fuller

March 1970

Prepared under

Contract Nonr-2342 (00)
Task NR 062-316

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OFFICE OF NAVAL RESEARCH
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BENJAMIN FRANKLIN PARKWAY • PHILADELPHIA, PENNA. 19103

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This report is the substance of a talk presented before representatives of ONR in Washington, D. C., on January 27, 1970.

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ABSTRACT

This report is the substance of a talk presented before representatives of ONR in Washington, D.C. on January 27, 1970. The review covers results of research in a spectrum of gas-bearing configurations, applications of gas bearings in various use categories, and future prospects.

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A REVIEW OF RESEARCH IN THE FIELD OF GAS-LUBRICATED BEARINGS

Introduction

This is to be a review of recent research in gas lubricated bearings. I will attempt to point out some of the problems, summarize some of the accomplishments and then consider some future prospects.

Apparently the first to suggest that air might be used as a lubricant was G. Hirn in 1854 (Ref. 1). No particular activity was recorded in this field until 1897 (Ref. 2) when Albert Kingsbury found that a close-fitting horizontal steam piston, when rotated, was supported by an air film. He built the first gas-lubricated journal bearing and proceeded to measure pressure profiles in the hydrodynamic film that was generated.

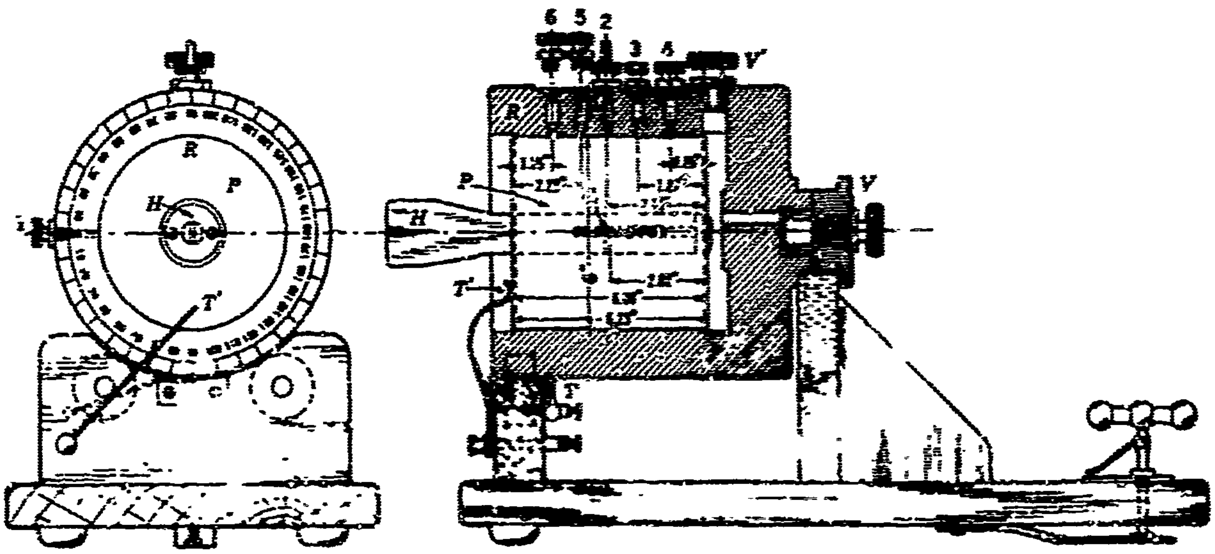
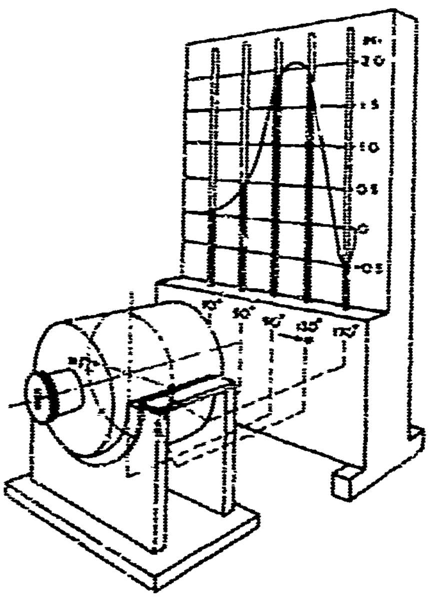


Fig. 1 Kingsbury air-bearing test apparatus.

The first theoretical study was that of Harrison in 1913. He included the effect of compressibility and applied the results of his analysis to the experimental data obtained by Kingsbury. (Ref. 3)

Stone in 1921 built an air-lubricated thrust bearing with a glass thrust collar and quartz shoes so as to permit the measurement of operating film thicknesses by optical interference bands. (Ref. 4)

In more recent times (1953), Drescher presented a significant paper on lubrication with gas. (Fig. 2)



Self-acting pressure generation in a journal bearing (Drescher)

Fig. 2

At that time with the exception of Harrison's work, the state-of-the-art consisted essentially of taking concepts developed for incompressible lubricants and extending them to air. There was no sophistication of analysis. It wasn't necessary at the time.

I became interested in the phenomenon during the 1940's, because of the obvious advantages. These are now well-understood but I might mention them briefly once again:

Cleanliness. Elimination of contamination caused by more typical lubricants.

Reduction and frequently the elimination of the need for bearing seals. Use of process fluid lubrication.

Stability of the lubricant. No vaporization, cavitation, solidification, or decomposition over extreme ranges of temperature from cryogenic (-450F) up to approximately 3000F.

Low friction and heating with no cooling generally required. Permits practical attainment of high speeds, (RPM well above 100,000 RPM).

I recall publishing a paper in 1947 that included something on hydrostatic, air-lubricated bearings (Ref. 6). In 1950 I gave a two-day seminar at Du Pont on the subject of air-lubricated bearings. In 1953 I published a paper on the low friction properties of air lubricated bearings (Ref. 7) and in 1956 in a textbook on lubrication a chapter was included on air-lubricated bearings (Ref 8).

In 1957 Captain Sawyer and Mr. Stanley Doroff of ONR organized a program of research in this field with The Franklin Institute. This was shortly expanded in a limited way to some other research groups like General Electric Research Laboratory, IBM and a few others.

The financial support, as I am sure you know was the result of a coordination effort by many agencies, pooling some of their resources for the support of the program.

I think it would be correct to say that by most standards the total amount of funds involved was rather modest but as you also know these funds were used very effectively and truly took on the role of seed money in the national and international research activity that resulted.

A major contributing factor to this effectiveness was a series of quarterly technical coordination meetings that was initiated by Captain Sawyer and Mr. Doroff. Attendance was by those groups supported, at least in part, by the cooperative effort administered by ONR plus, by invitation, those who were known to be working in the field of gas-bearing research.

The meetings were kept small, very informal and friendly so that the interchange of ideas, suggestions and criticisms has proven to be very helpful and most constructive. A large amount of duplication of effort that would normally be expected has been avoided. Many individual research efforts have been speeded on their way to a constructive conclusion as a result of these meetings.

I might add that just two weeks ago a technical coordination meeting was held at the Instrumentation Laboratory of Mass. Inst. of Technology with 39 in attendance, with 17 presentations, plus general discussion.

It is interesting to observe that in the United Kingdom this

organizational pattern has been emulated by Mr. Henry Elwertowski of the Admiralty Compass Observatory, with a group of Britishers conducting research in the gas-lubricated bearing field. There is considerable cross-communication between the U.K. group and the U.S. group, culminating in a joint meeting, once a year. Last April (1969) this combined meeting was held at Columbia University. Next April it is scheduled for England.

<u>What is a gas-lubricated bearing? Difficulties of analysis.</u>

The gas-lubricated bearing is typical of fluid film bearings using liquid or semi-liquid lubricants with the exception of course that the lubricant is a gas. This complicates both the analysis and the behavior of these bearings because of the compressibility effects of the gas and because of the absence of significant damping that normally exists with viscous liquid bearing films.

It is interesting that because of damping in liquid lubricated bearings, many, if not most of the possible modes of vibration and critical speeds of rotors were suppressed. The dynamic analysis and critical speed analysis of rotor systems was rather fuzzy in that it could analyze the problems that presented themselves but wasn't usually sharp enough to explore dynamic behavior patterns that had not been observed experimentally. With the advent of gas lubricated bearings this was all changed drastically.

For example we had at one time a rather simple rotor supported

on two externally-pressurized gas lubricated journal bearings and one double acting, gimbal mounted thrust bearing. This rotor and its bearings exhibited something like 12 or 13 possible resonances.

The whole field then of rotor dynamics has had to be carefully and rigorously re-examined. We now think we understand the multiple resonant conditions that may exist in any bearing-supported rotor system and as a result have a much greater appreciation of the benefits of liquid film damping in suppressing or controlling many of these conditions.

The compressibility of the film and its time transient behavior has had to be fully explored and understood. As Captain Sawyer once phrased it, we have had to initiate a study of compressible fluid flow in very small passageways. We have had to deal with what might be called "micro aerodynamics." And that is how it all began --- with a study of the properties of a simple, compressible fluid film under static load conditions. Once this was understood, the next step was to start applying these general analyses to typical simple bearing geometries.

I have enclosed a list of the reports published by only one group engaged in research on gas lubricated bearings, The Franklin Institute, as a typical illustration of how the program has developed. (This list will be found in the Appendix).

Notice that the early reports, specifically Nos. 2 and 5 are basic to an understanding of the compressible fluid film phenomena. These were established first before considering more complicated geometries. The measure of the degree of compressibility was established by the parameter Λ which was defined for a journal bearing as:

$$\Lambda = \frac{6\mu\omega}{p_a}\left(\frac{r}{c}\right)^2$$

where ω= angular velocity of the journal, (rads/sec.)

r= radius of journal (in.)

c= radial clearance (in.)

p_a= ambient pressure (psia).

With small values of compressibility number, Λ, approaching zero, the film behaves as though it were incompressible and the corresponding equations from the previous literature on liquid lubricants may be used.

For extremely high speeds, or very low ambient pressures, Λ is high, and compressibility effects are a maximum. A number of solutions have been developed for the limiting case of $\Lambda \to \infty$.

Analysis of compressible gas films is at best rather difficult. This can be demonstrated by an examination of the basic Reynolds equation for a gas. When the film is a gas, this equation is non-linear in the pressure. The Reynolds equation is derived by combin-

ing the Navier Stokes equation, the continuity equation, and the equation of state for a perfect gas.

$$\frac{\partial}{\partial x}\left[h^3 p \frac{\partial p}{\partial x}\right] + \frac{\partial}{\partial z}\left[h^3 p \frac{\partial p}{\partial z}\right] = 12\mu \frac{\partial [ph]}{\partial t} + 6\mu \frac{\partial}{\partial x}\left[p h \left(U_h + U_0\right)\right] + 6\mu \frac{\partial}{\partial z}\left[p h \left(W_\ell - W_0\right)\right]$$

(ISOTHERMAL)

The two salient features of the Reynolds equation are the linearity in the pressure p and the time diffusion term $\frac{dp}{dt}$. These are different from the Reynolds equation for incompressible lubricants and introduce complications.

Computer techniques.

In the analytical area I think it would be agreed that truly significant advances have been made in the handling of digital computer solutions. Besides yielding solutions for the gas-lubricated bearings of interest, these techniques have been worthy contributions in their own right to the general field of numerical analysis. Castelli and Pirvics have published a summary of this work (Ref. 9).

Fig. 3 is taken from this report and summarizes in tabular form the many techniques that have been developed and are used in the field of gas-lubricated bearings. A constant objective has been to reduce computer running time, increase speed and accuracy

of asymptotic convergence and force numerical stability. Considerable success has been achieved so that these techniques are finding use in other fields of engineering and science.

As one can see there are two basic avenues of approach, the direct numerical and the alternate is the analytic-numerical. Castelli says that in general the direct numerical is probably cheapest now with the large memory, high-speed computer. It can be used effectively in so-called abnormal problems whose boundary layer effects may arise at high values of compressibility number Λ .

This technique approaches the problem immediately with numerical approximations, the accuracy of which is often easier to control. Advantages are: less algebraic involvement, no limiting assumptions besides those imposed by truncation errors. Disadvantages are: increased amount of computer time, difficulty in discerning parametric trends, the need for an efficient program.

The other approach is the combination, analytical-numerical. Some approximations are made in order to partially solve the problem by analytical means. This type of treatment usually terminates with a numerical computation less demanding in computer time than the first. Among the advantages are increased ease of parametric studies and possible reduction in overall cost of execution.

Among the disadvantages are the fact that approximations lead to limited or uncertain ranges of applicability and that the amount

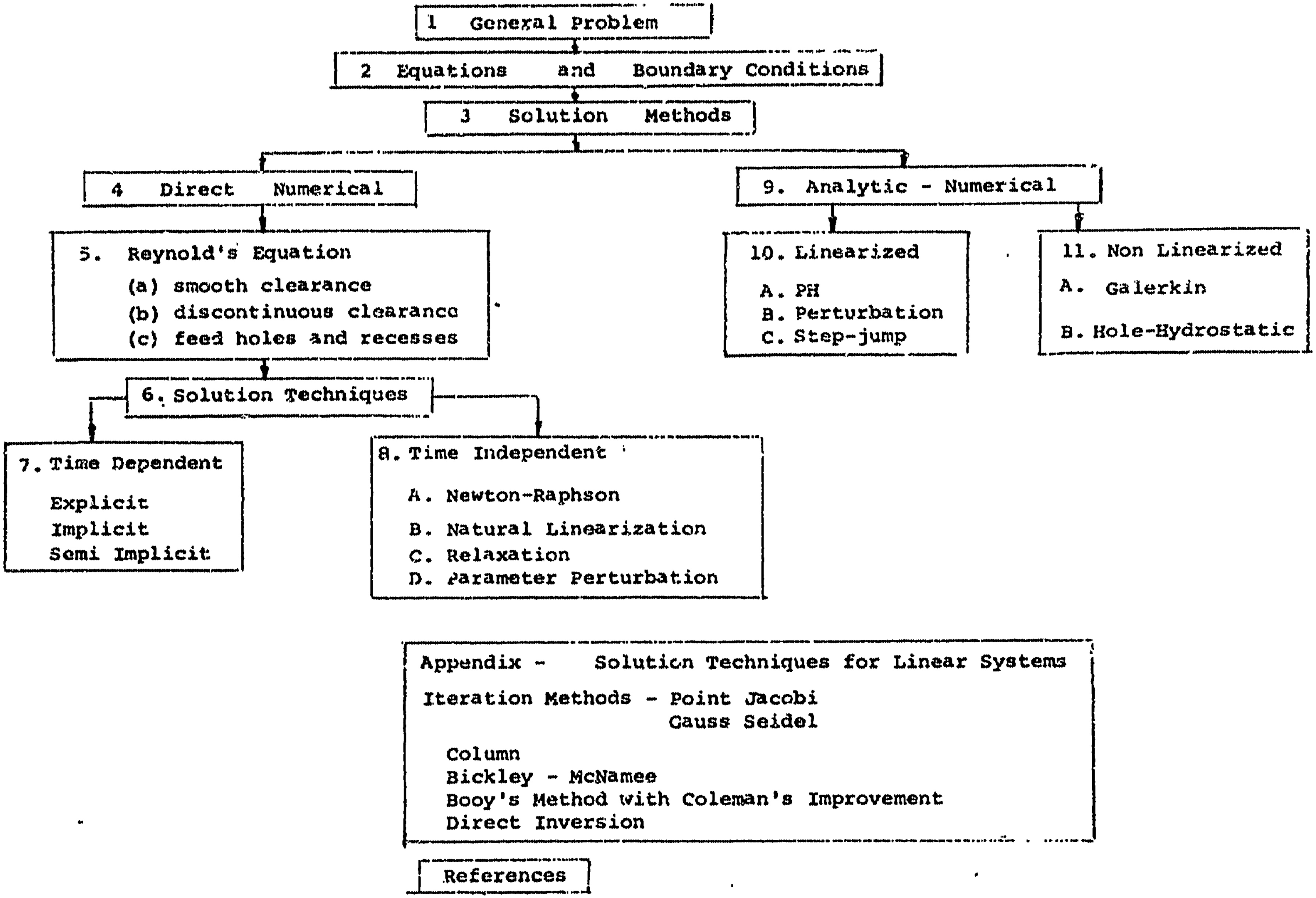


Fig. 3 Logical organization of report.

of algebra to be dealt with is greatly increased which always leads to the probability of incorporating a systematic error.

The time dependent solutions involve the integration of Reynolds equation including the variation in time of the pressure distribution dp/dt as well as the response of the motion of the film geometric boundaries dH/dt. It is commonly employed to simulate the behavior of dynamic systems.

Such problems require the simultaneous integration of the appropriate equations of motion, governing all dynamic degrees of freedom.

Time independent solutions. Probably the most commonly encountered problem in lubrication technology is that of producing data for equilibrium operation of gas bearing systems. Sometimes the steady state solutions are obtained as asymptotic limit solutions of time dependent analyses. However, it is usually more economical to use the Reynolds equation with the time derivative terms in pressure and film thickness, set equal to zero.

Possible exceptions would be when the equilibrium conditions of the system are not known beforehand such as for example with tilting pad bearings with assigned pivot position and pivotal clearance, and for any bearing system for which the geometric position corresponding to a few externally applied specified loads is desired.

Since the Castelli paper was published in 1968 a new technique has been developed for gas-lubricated bearings, having been borrowed from the field of structural analysis. It is called the finite element programming technique. Where sharp changes in pressure are encountered in gas bearings, a finer grid mesh is necessary to ensure convergence. However if the entire bearing is treated with a close grid spacing more computer time and a greater production cost would be required to get a solution. The finite element technique has proven to be very useful in being able to handle variable grid spacing. It also readily handles large systems of equations and has been used effectively on both incompressible and compressible lubrication regimes.

Two papers by Dr. Reddi of The Franklin Institute were presented at the October 1969 Joint Lubrication Conference of ASME-ASLE on the use of this technique in solving fluid-film lubrication problems (Refs. 10 and 11).

Results of research effort.

Now what have been some of the results of these endeavors?

We see that the analysis of static loads on gas-lubricated journal and thrust bearings is well in hand. Quite a spread of bearing types has been investigated. In general a level of understanding has been achieved that has permitted publication of design curves for several of these bearings. I will mention a few examples.

Plain journal bearing.

Fig. 4 shows a typical representation of a complete, 360 degree journal bearing (Ref. 12). Notice the eccentric distance e between the center of the bearing 0 and the center of the journal 0'. The ratio of this distance to the machined-in radial clearance is called the eccentricity ratio ε which will of course vary between 0 for the concentric case where 00' = zero, to the other limit of $\epsilon = 1$ where 00' = the radial clearance.

Also notice the angle $\emptyset$, the attitude angle. This is the angle between the line of centers and the direction of load application.

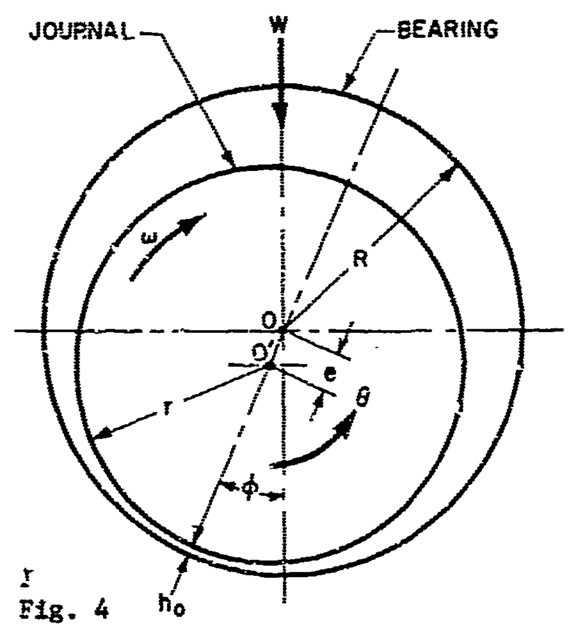
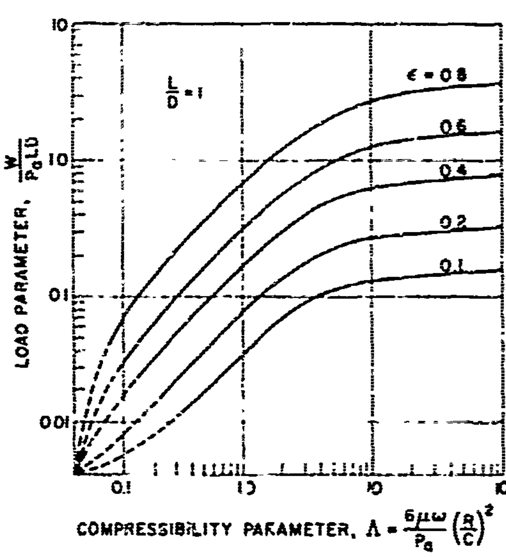


Fig. 4

Journal in 360 deg bearing

Fig. 5 shows a typical determinationless load plotted against the compressibility parameter Λ for several values of eccentricity ratio ϵ. Notice that the entire plot is for a length to diameter ratio $(\ell/d)=1$ (Ref. 13).



Theoretical load-carrying parameters versus compressibility parameter for full journal bearing for $l/d = 1$

Fig. 5

Fig. 6 shows a comparison between the computed load-carrying capacity for a given bearing using incompressible fluid solutions as compared to compressible fluid solutions (Ref. 14). Notice that for gases, because of compressibility effects the load capacity does not increase directly with speed but reaches as asymptotic value at high Λ. There is essentially no increase in load capacity for example in Fig. 6, between conditions where $\Lambda = 100$ and $\Lambda = 1000$. This is unlike the behavior of liquid film bearings.

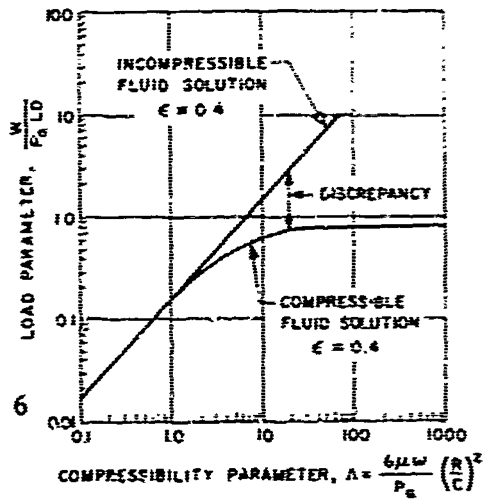


Fig. 6

Discrepancy in load capacity between theories based on incompressible and compressible lubricants

Attitude angle.

Recall now the angle marked ϕ in Fig. 4. This is called the attitude angle and is the angle between the line of centers OC' and the direction of the applied load. In Fig. 7 this angle is shown again as applied to the location of the journal center.

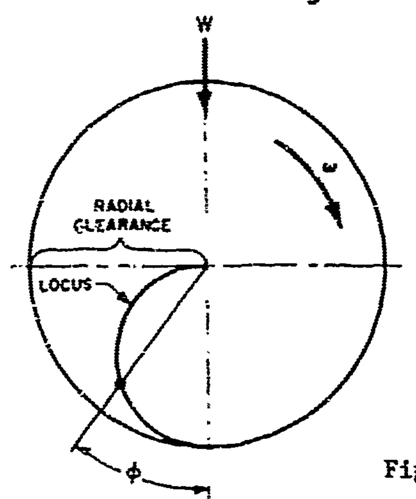


Fig. 7

Typical attitude—eccentricity locus for the motion of the center of a journal in the clearance of a self-acting bearing

The curve marked "locus" designates the path followed by the journal center within the clearance space as the load and/or the speed varies. It is shown more precisely in Fig. 8

where one quadrant of the clearance circle is used to plot the attitude-eccentricity locus.

This angle is important in establishing the degree of stability of a shaft in a journal bearing. For gas lubricated journal bearings the locus path and attitude angle are a function of Λ as well as ϵ. Fig. 9 shows this relationship. In general, the smaller the attitude angle the greater the stability of the journal in the bearing. It can be seen in Fig. 9 that as Λ increases, the attitude angle ϕ decreases. This does not happen with incompressible lubricants. Thus we see this adds another degree of complication to the analysis of stability of gas lubricated bearings.

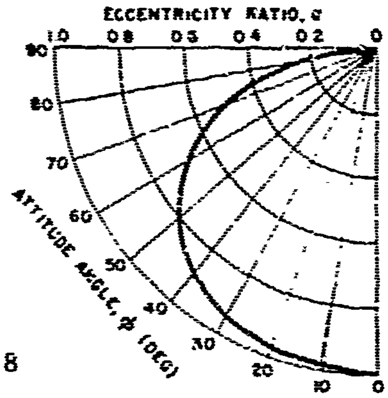
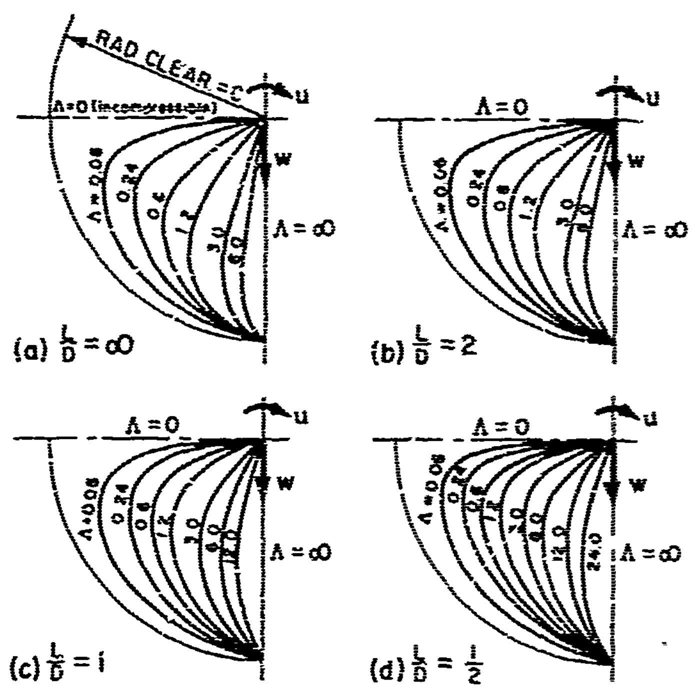


Fig. 8

Attitude—eccentricity locus for liquid-lubricated 180 deg journal bearing



Attitude—eccentricity locus diagrams for full, gas-lubricated journal bearings

Fig. 9

Grooved bearings.

Various geometric modifications such as slotting or grooving aid stability by generally reducing the attitude angle for a given operating condition. Fig. 10 shows an axially grooved

journal bearing where one or more grooves may be used. A common number is three. Carrying this to the limit we have the herringbone grooved journal bearing Fig. 11 (Ref. 15) and the Rayleigh step bearing Fig. 12 (Ref. 16). Both of these types are very much more stable than the plain journal bearing.

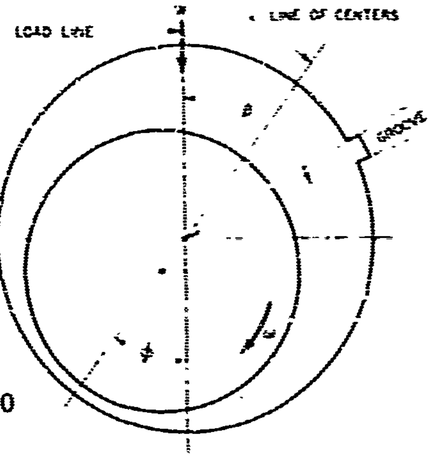
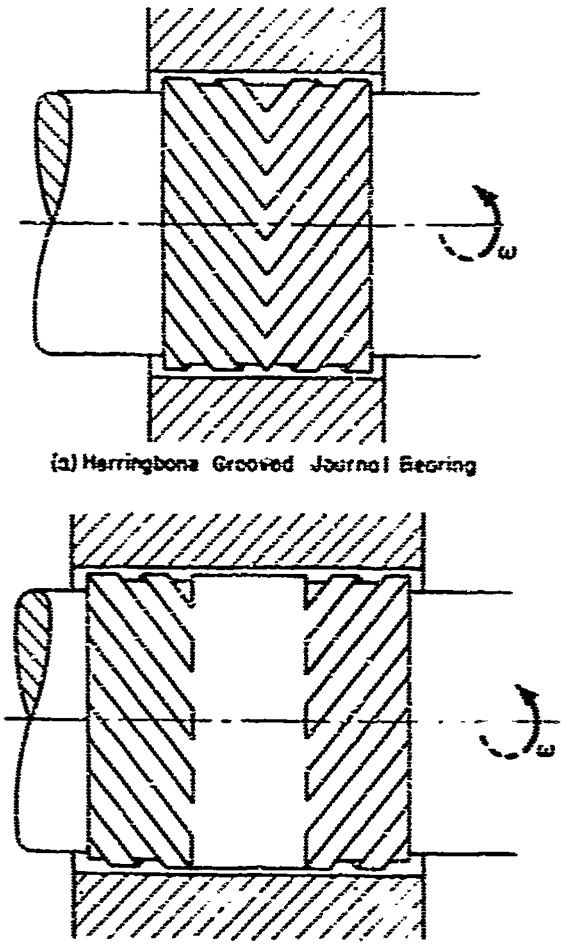


Fig. 10

Axially grooved journal bearing

<u>Tilting-pad journal bearing.</u>

The ultimate in stable operation however for conventional bearings is the tilting-pad journal bearing. Fig. 13. This type of bearing can have three or more shoes (Ref. 17). Although

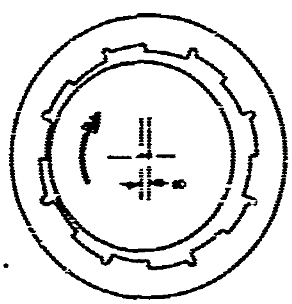


(a) Herringbone Grooved Journal Bearing

(b) Journal Bearing, Partly Grooved; Symmetrical Pattern

Herringbone groove patterns

Fig. 11



Rayleigh step journal bearing

Fig. 12

mechanically more complicated than the fixed geometry bearing, it is finding wide use in high speed rotor systems with both gases and liquids. Considerable effort has gone into the analysis of these bearings because of the need to understand their behavior

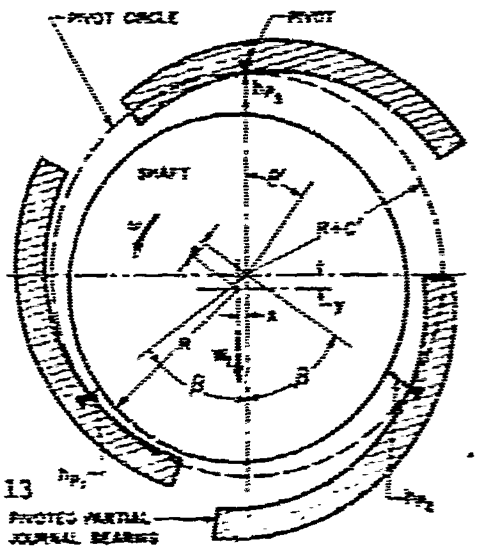


Fig. 13

[illegible]

with so many possible degrees of freedom. Besides permitting the shaft to move in a variety of ways, the shoes themselves can pitch, roll and yaw and if one shoe is spring loaded it may also translate in a radial direction. Thus a 3 shoe journal bearing will have 10 degrees of freedom just in the bearing alone. The shoe motion is determined in part by the mass of the shoe and the way it is distributed and by the friction or lack of it in the shoe pivot. The preload on the shoes has also a sensitive influence on the bearing behavior. Fig. 14 shows a typical field map for a three pad tilting-pad journal bearing of a particular configuration.

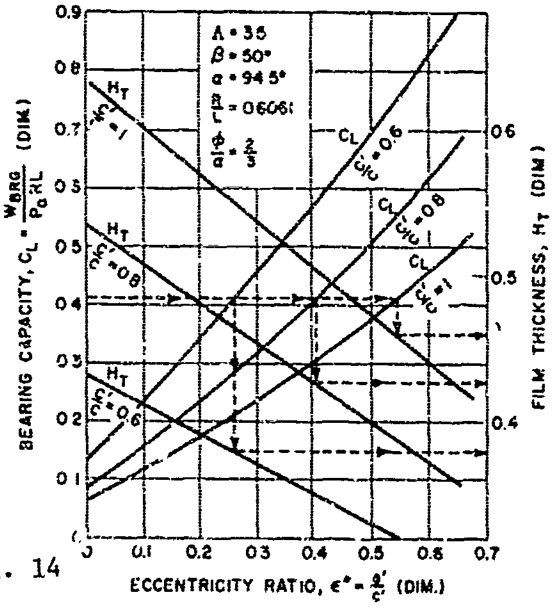


Fig. 14

Total bearing load coefficient for three pads versus eccentricity ratio and trailing edge film thickness

Details are shown on the slide.

Notice that with higher preload, $\left(\frac{c'}{c} = \frac{\text{Actual}}{\text{Machined}}\right)$ the greater the load-carrying capacity for a given eccentricity ratio. However, the trailing-edge film thickness is diminished somewhat with an increase in preload.

Instabilities.

Instabilities in self-acting journal bearings are of paramount concern and much of this presentation has so far been devoted to that concern.

Two kinds of dynamic instabilities may be found in fluid film bearings whether liquid or gas lubricated. As a result of the low damping properties of the gas film, these instabilities in gas-

lubricated bearings become more prominent than with liquid-lubricated bearings. The first kind of instability is associated with typical spring-mass natural frequencies where the bearing fluid film is the spring. When the operating speed corresponds to one of these natural frequencies, we have a so-called "critical speed." As with usual spring-mass resonances it is possible to have stability on either side of the critical.

The second kind of dynamic instability is a self-excited vibration characterized by having the center of the shaft orbit around the center of the bearing at some frequency equal to or somewhat less than one half of the spinning or rotational velocity of the shaft. Under these conditions it has been shown that the capacity of the bearing to support radial load is sharply reduced and may fall to zero. In Fig. 15 this orbital motion is depicted where Ω is approximately equal to 1/2 of ω. The orbital velocity is Ω and the spin angular velocity is ω.

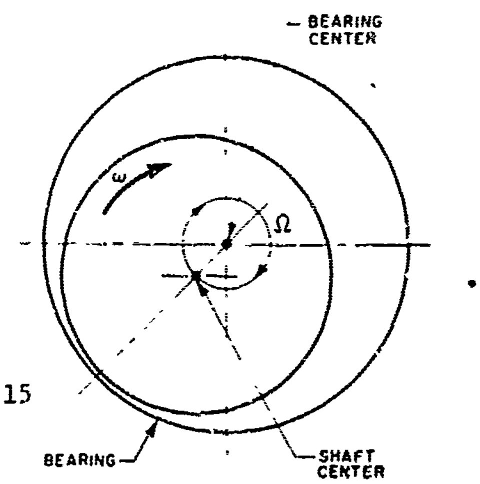


Fig. 15

Orbital whirling velocity of shaft center Ω about center of bearing compared to shaft spinning velocity ω

The shaft system may be stable as the speed is increased until the threshold is reached. Crossing this threshold by further increase in speed will bring the system into a region of instability, which becomes more violent as the penetration becomes deeper, until inevitable seizure results. Unlike an ordinary critical speed, the shaft cannot pass through this one and attain a region of stability on the other side at a higher speed.

Rotor dynamics.

Returning now to instabilities of the first kind, the typical critical speeds, we find them referred to in the literature as synchronous critical speeds, since they are synchronous with some shaft disturbing rotational speed. They may be of a translational form or of a conical form although the relative magnitude of the transverse moment of inertia of the rotor, as compared to the polar inertia of the rotor, and its mass, can make one form predominate over the other.

In general the quasi-static, equilibrium value of the spring rate of the film is coupled with either an appropriate mass or the moment of inertia of the shaft to determine a natural frequency.

If the bearing is freely or flexibly supported (as contrasted to the journal) it also may have a resonant translational frequency or conical frequency. If the bearing is freely supported and rotating, then like the journal, the polar moment of inertia and

gyroscopic effects should be considered.

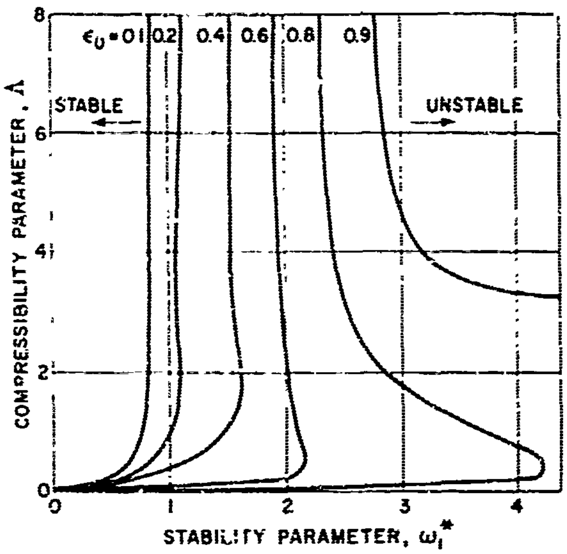
The instabilities of the second kind are much more difficult to analyze and evaluate. These are the self-excited forms of dynamic response. There have been numerous attempts to achieve a satisfactory solution but not until recently has some conclusive theoretical work been performed. Some of the early attempts neglected the dP/dt term and were therefore limited to very low values of Λ and ϵ.

Of the several solutions now available I will mention only two of those developed by Castelli and Elrod (Ref. 18). They have included the time-dependent (history) terms and have provided two solutions to the self-excited instability problem.

In the first, nonlinearities are eliminated by restricting the analysis to possible shaft motions within a very narrow range about an equilibrium position. This is called the small perturbation method.

The second method includes the complete nonlinear equations which are integrated numerically to obtain the shaft-center orbits corresponding to any set of geometrical, running and initial conditions. The technique essentially uses the high-speed digital computer as a very accurate experimental rig. It operates exactly in accordance with the assumed governing equations. This is called thw orbit method. The path of the journal center is followed as it runs on the computer. If the locus spirals outward with an

increasing radius, the condition being predicted is unstable. Conversely if when disturbed the locus returns to a central equilibrium position, the bearing-rotor system is stable.



Plot of half-frequency translatory whirl threshold for infinite length 360 deg journal bearing

Fig. 16

Considerable computer time is involved with this approach so that it is expensive to determine complete stability maps. However some significant results have been obtained as shown by Fig. 16. A principal value of this method is to check on the accuracy of approximate methods for determining stability thresholds of gas-lubricated journal bearings.

The stability parameter $\omega_1^* = \omega\sqrt{\dfrac{CM_1}{W}}$

where ω = shaft speed (rads/sec.)

c = radial clearance in the bearing (in.)

M_1 = mass per unit length $\left(\text{lb. sec.}^2/\text{in.}^2\right)$

w = load per unit length (lb./in.)

The value of eccentricity ratio for a bearing of infinite length is designated as ε_o . If the intersection of ω_1^* and Λ falls to the left of the curve for the operating value of ε_o , the bearing should be stable. If it falls to the right the bearing should be unstable.

Corresponding efforts have been made to probe into the instabilities of tilting-pad journal bearings but because of the many degrees of freedom they are necessarily more involved and complicated. As indicated earlier one of these bearings can have 10 degrees of freedom. The rotor itself has 2 degrees of synchronous freedom, translational and conical, and also 2 degrees of self-excited whirl again translational and conical. This means then that a relatively simple, two-bearing rotor system can enjoy 24 degrees of freedom. With a thrust bearing, additional degrees of freedom would be added.

The number of individual bearing parameters that would be involved could easily amount to 20 when all of the masses, moments of inertia, and film stiffnesses are considered.

It should be said again the effort expended in this area of gas-bearing rotor dynamics has resulted in a greatly expanded understanding of rotor dynamics in general. One might even say that there has been a real "break-through" in our ability to deal with rotor critical speeds and rotor dynamics over their entire spectrum. This is significant step forward in the development of our technology.

Leaving the subject of rotor dynamics now, I would like to mention a few other types of journal bearings that have been examined in some detail as a part of the general research program on gas-lubricated bearings.

Vapor lubrication.

The first of these would be vapor lubricated bearings. In line with the practice of using conveniently available process fluids as bearing lubricants, vapors present themselves as candidate lubricants. The behavior of vapors in bearings should be quite different from that of gases because of their tendency to partially condense under the right conditions of pressure and temperature. This would produce a two-phase lubricant of combined liquid and vapor.

Unterberg and Ausman (Ref. 19) show that thermodynamic considerations prescribe that the temperature remains constant throughout the bearing. When the maximum pressure in the bearing film reaches the saturation vapor pressure at that constant temperature, a further increase in bearing load then causes partial condensation instead of a rise in maximum pressure as one would normally expect with a single-phase lubricant. Fig. 17 shows some numerical results for the case of $\Lambda = 1$. This is a plot of dimensionless load capacity versus eccentricity ratio ε. The parameter on this graph is saturated vapor pressure p_s, divided by atmospheric pressure p_a.

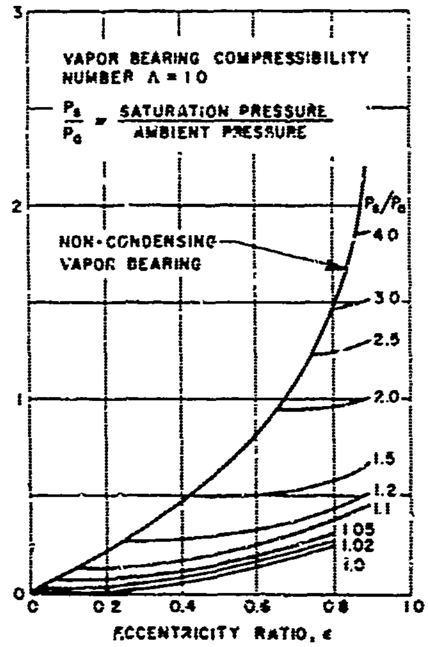


Fig. 17

Load capacity of long isothermal condensing vapor journal bearing

If ε_{cr} is exceeded for a given p_s/p_a , the condensing regime begins and the load capacity falls below that for the noncondensing gas. Notice that in the condensing region of the bearing, as the ε increases the load-capacity does not increase very much. Thus there is a distinct danger of film collapse if an increment of load is added to that on the bearing.

Since partial condensation in a self-acting, vapor-lubricated bearing essentially weakens the bearing and reduces its load-carrying ability, recent attention has been focused on the externally-pressurized type.

Externally-pressurized bearings.

This brings us to the externally-pressurized form of gas-lubricated bearing. This bearing has the ability to support a

load independent of rotation. Fig. 18 (Ref. 20). Thus at zero speed the full load carrying capacity of the bearing can be developed. This is coupled with the highly desirable phenomenon

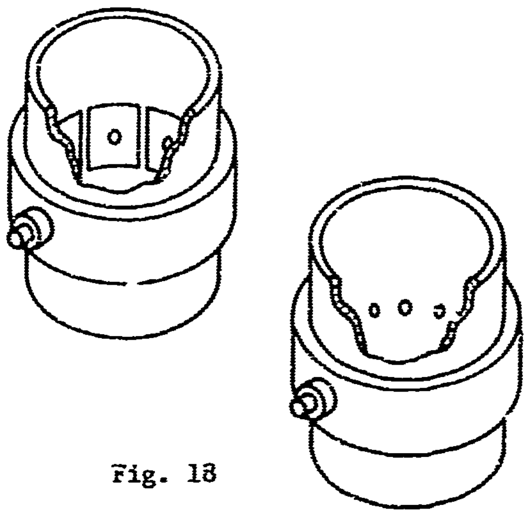


Fig. 18

Hybrid journal bearing

that at zero speed in a properly designed and manufactured externally-pressurized bearing, the friction is also zero. These two aspects provide great latitude to the bearing designer and in conjunction with a controlled film thickness and also film stiffness, provide him with capabilities not available from any other type of bearing.

Many significant accomplishments have been achieved in the application of externally-pressurized bearings to instrument bearings such as for gyroscopes and to high speed dental drills and orthopedic grinders. They have been used in what has been called hybrid bearings where the external-pressurization supplements the

self-acting film pressurization of the bearing. Thus the rotor can be lifted and supported on a gas film before rotation begins. This eliminates the high-friction starting condition and also eliminates any possibility of scoring or abrading the surfaces.

The external pressurization can also be used to suppress self-excited whirl instabilities in a journal bearing or to change the critical speed of synchronous whirl.

The subject is very broad. Gross, for example in a recent survey paper on the subject (Ref. 21) lists 17 different types of externally pressurized journal bearings and 45 different forms of thrust bearings depending upon the geometry and the type of flow restrictor employed.

Air hammer instability.

As one might expect there is a very serious dynamic problem in these bearings and if care is not taken in the analysis and construction one may end up with a pneumatic vibrator or a pavement breaker instead of a bearing.

Licht and Elrod (Ref. 22) have provided the most thorough study of air-hammer instability, although it was restricted to a simple circular thrust bearing. They were able to show that this instability is a complex phenomenon involving the nonlinear interaction of different bearing characteristics. It was shown that by varying the bearing parameters, the bearing can go into or out of

pneumatic instability at a range of different conditions. For example, there may be air hammer at intermediate film thicknesses but none at larger or smaller film thicknesses. Unfortunately the analysis of this phenomenon is complex and simple approximations do not seem to exist.

For all but very simple geometries the design procedure is necessarily limited to one of trial-and-error although certain guide lines and recommendations are available.

There are four principal types of compensation employed in externally pressurized bearings. These are the orifice, the capillary, the porous plug and inherent compensation. The first three describe the type of restrictor placed between the bearing and the gas pressure source. Inherent compensation means that the resistance to flow into the bearing occurs at the film inlet. Under these conditions the minumum area through which the incoming gas must flow in effect forms an orifice. The inherently compensated bearing is the most resistant to air hammer while the capillary compensated bearing is most likely to be unstable. Orifice and porous plug bearings have performance characteristics somewhere between these limits. One major problem with orifice compensation is the possibility of orifice closure by accumulated dirt particles. Porous plugs tend to act like filters and over a period of time will collect particles and become clogged. There is also

a problem in machining sintered porous material so as to avoid clogging the fine pores.

Present design techniques are essentially limited to laminar flow in the film clearance space. Several attempts have been made to examine the shock waves and supersonic flow regimes that exist in these bearings especially if the supply pressure is relatively high and the films relatively thick. Correlation so far, for design purposes, has not been satisfactory.

The configuration of pressure sources is also limited at the present time to the simplest geometries.

<u>Precision manufacture.</u>

For these bearings and all the other types that have so far been mentioned the questions of precision manufacture, and maintenance of design film thickness and alignment are of vital significance. It has been said that gas-lubricated bearings are less forgiving than oil-lubricated bearings. They are less forgiving of errors in estimating loads or of deviations from specifications during manufacture and installation and less forgiving of distortions that may find their way into the rotor, the bearing components or the housing. This has been a real problem in many ways but especially with the ultra-precision gas bearings of gyroscopes. Radial clearances in these journal bearings may be as small as 50

to 75 micro inches or less and to meet performance specifications these clearances must be held to a high degree of tolerance during the life of the instrument.

<u>Mechanical design.</u>

Two papers on this aspect of bearing design are I think especially significant. One is by Schetky (Ref. 23) and the other by Maringer (Ref. 24).

Schetky examines some of the causes of dimensional changes and indicates that they are the result of relaxation of internal stresses producing micro-creep, thermal expansion, plastic flow and metallurgical changes.

He says a high degree of stability can be obtained by following good practices while forming or machining the part. This is to relieve the stresses set up at each step of metal forming or cutting. A typical sequence could consist of nine steps. For example:

1. Stress relieve or anneal
2. Rough machine
3. Stress relieve
4. Heat treat for desired physical properties
5. Semi-finish machine to a slight oversize
6. Stress relieve

7. Finish machine, grind lap or hone

8. Stabilize by thermal cycling or heat treatment

9. Final lap or hone

The stabilizing process will vary with the alloy. The simplest is to hold the part at 200F for 24 hrs. or longer. This isothermal treatment is easy to do especially if the parts are large.

When the part is subjected to subzero temperatures either in storage or in service, a temperature-cycling treatment is recommended. This is a stabilizing step that induces the residual stresses to relieve themselves. Cycling is done from say 75 to 212F down to a subzero immersion in a bath of dry ice and acetone, or dry ice and alcohol, typically - 100F. The cycling produces a more stable material.

Schetky shows that a 1020 annealed steel after 10 cycles from 200F to - 100F, can show a dimensional change of 15 microinches/in.

303 stainless (quench-annealed) after 10 cycles can show a change of 40 microinches/in.!

For parts subjected to load, the influence of elastic and plastic strains must be included. For best dimensional stability, metals should be stressed below their elastic limit. But where is the elastic limit stress?

A true elastic limit with no plastic strain is not easy to identify. Probably the best elastic limit presently available is

one that identifies a residual plastic strain of one microinch/in. This is called the micro yield stress (MYS). This "elastic limit" can be as low as 1/4 of the usual 0.2% offset yield strength and drops even further with only a slight increase in temperature.

For example Schetky quotes a 52100 bearing steel as having a room temperature yield stress (0.2% offset) of over 250,000 PSI. However this same material has a micro-yield stress of only 87,000 PSI. Remarkably, by increasing the temperature to only 165F the micro-yield stress falls to 30,000 PSI. Being unaware of this, the designer could easily have the bearing subjected to stress levels reaching into the plastic regime, with subsequent loss of precision dimension.

It is surprising to learn, through this work, that the modulus of elasticity is also temperature dependent and is also stress dependent if one is thinking of precise dimensional stability. Schetky shows a change in the elastic modulus E of up to 8% for commercially pure aluminum over a range of only 12,000 PSI.

Temperature dependence of E also shows up even for a small range of from - 50F to 150F. Variations in E are listed as 6.5% for 2024-T4 aluminum, 2.9% for a 1090 steel and 4.5% for an 18-8 stainless steel for this same small temperature range.

It is interesting to see that the coefficient of thermal expansion can also change as much as 3% over a <u>stress range</u> of

12,000 PSI. This is another aspect of metallurgy and of physical constants of materials that is frequently overlooked.

Micro creep.

Considering an SAE 52100 bearing steel, with an applied stress of about 100,000 PSI, the creep after 100 hours will be as follows:

Temperature	Creep	Micro yield stress
35F	20 in./in. x 10^6	95,000 PSI
95F	50 in./in. x 10^6	87,000 PSI
165F	400 in./in. x 10^6	30,000 PSI

It is understood that the micro yield stress at these three temperatures is 95,000 PSI at 35F, 87,000 at 95F and 30,000 at 165F.

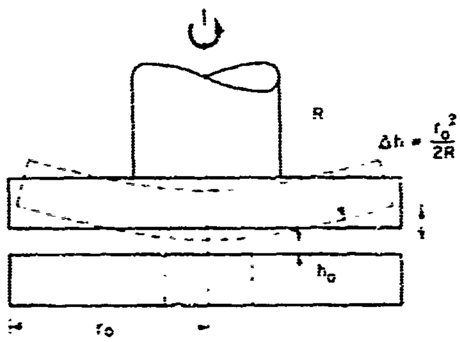
Maringer shows that at room temperature, metal will creep even if held at stress levels below the micro yield stress.

Prestrain raises the conventional yield properties but surprisingly prestrain greatly reduces the microyield properties. It is imperative that designers become familiar with these facts.

The need for precision dimensional tolerances in gas-lubricated bearings has provided a stimulus to the investigation of these matters, for the ultimate benefit of design in general. For example Maringer has presented two papers on the subject within the last two years and has been encouraged to prepare a book (now in process) for the guidance of designers.

COMPLIANT SURFACE BEARINGS

Under some circumstances, in spite of a thorough understanding of physical properties and in spite of recommended manufacturing procedures, the physics of a particular design may introduce almost insurmountable difficulties if one is restricted to the use of rigid surface bearings. For example, Fig. 19 shows the distortion of thrust plates in a spiral grooved thrust bearing due to heat generation in the fluid film (Ref. 25). A simple solution would be to have one of the mating surfaces compliant in some way so that it might match the curvature induced in its opposing member and thus be able to maintain a uniform film thickness.



Distortion of thrust plates due to heat generation in the fluid film

Fig. 19

Foil bearings.

All of the bearing types that have been described so far in this talk have been correctly assumed to have rigid surfaces. They were certainly rigid as compared to the stiffness of the gas film. However, there are fields of application for bearings whose surfaces are flexible or compliant rather than rigid. The foil bearing is

the best-known of these, Fig. 20. It consists of a thin strip of flexible material such as plastic tape, thin metallic foil or the like contacting a simple journal as shown. As the journal spins a reasonably large force F can be supported by the self-acting gas film in the contact area between the tape and journal.

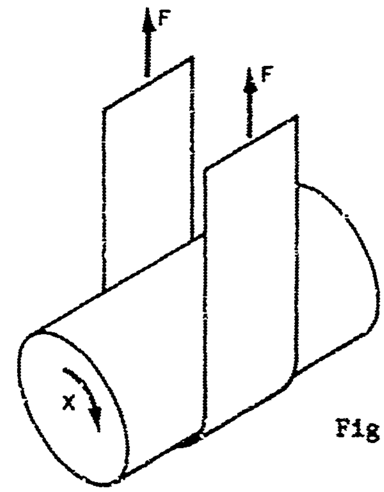


Fig. 20

Basic foil bearing arrangement

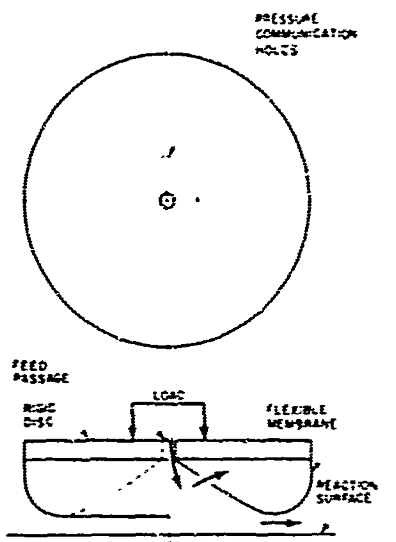
This can be used to apply a load to a rotating shaft in a very simple manner. However the greatest value of the concept is by far in tape transport for high-speed magnetic tape recorders, and in tape transport for digital computers. Here the journal is stationary and contains the recording or the read-out components while the tape glides past. Analysis of a foil bearing is complicated by the fact that the film thickness cannot be simply specified. It depends upon the self-generating pressure of the gas, the tension in the foil, the speed, the elasticity of the foil and the axial width of the foil.

Anticlastic curvature will also be present so that the film thickness at the edges of the foil will be smaller than in the center. However, these problems are being dealt with on both theoretical and experimental levels and considerable design information is presently available.

The foil bearing is by no means limited to mechanical input-output in electronic data processing. It can often be an essential part of continuous manufacturing of plastic film, metal foil, paper and textile materials. And most recently the foil bearing also holds out promise as a self-aligning stable support for high-speed rotors. Active research is continuing in this field.

Wildmann has recently published a summary paper (Ref. 26) dealing with foil bearings in which he includes 12 references, 11 of which were published since 1965.

Mention should be made here of the flexible membrane hydrostatic air bearing by Levy and Coogan, Fig. 21. This is a very effective load supporting device and is finding rather wide commercial use in manufacturing plants to support and transport pallets with vanishingly low friction over typically rough factory floors (Ref. 27).



Flexible membrane hydrostatic air bearing

Fig. 21

Load-carrying devices somewhat related to these designs are being considered for high-speed transport at speeds above those considered acceptable for the typical wheel-on-rail supporting device, i.e., above 150 MPH.

Castelli and Pirvics have examined the behavior of the elasto-hydrostatic, gas-lubricated circular thrust bearing (Ref. 28). This refers to a typical thrust bearing with one element of the pair of surfaces coated with a compliant elastomer layer. The results show an increase in load-carrying capacity over that of a rigid surface bearing for the same supply pressure. Flow rates are also decreased because of the compliant-surface action.

The compliant action of surfaces is now understood to be a dominant phenomenon in the lubrication of human joints containing synovial fluid.

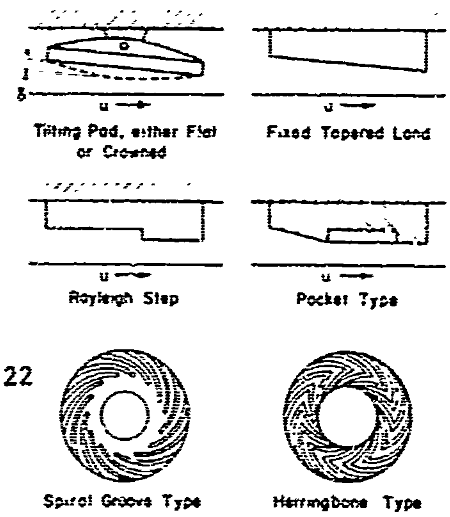
In a widely divergent problem the basic phenomenon of "hydroplaning" of automobile and aircraft tires on wet roadways and runways is due to the compliant action of the tread surface of the tire.

Obviously one of the inherent behavior characteristics of the compliant surface bearing is to accept misalignment and to conform to typical distortion of the bearing package. McCabe of The Franklin Institute has built a four foot diameter spherical bearing for an instrument package tester. Pan of M.T.I. has applied a compliant layer to a particular gyro design where when pressurized the compliant layer permits free rotation but when not pressurized, grips the rotor and acts as a brake.

Thrust bearings.

Looking back over the material presented in this talk, it will be noticed that thrust bearings have been mentioned only peripherally. The omission is only the result of lack of time. Gas bearing research efforts have been applied to thrust bearings with about the same level of effectiveness as for journal bearings. Fig. 22 (Ref. 17)

Fig. 22



Typical forms of self-acting thrust bearings

Squeeze bearings.

I have also made no mention of squeeze film bearings because of lack of time. These have also been the object of much interest and research effort.

Applications.

Now what has resulted from all of this? What are some of the applications of gas-lubricated bearings that can be referenced? There are many. And they are related to the advantages that were spelled out at the beginning of this talk, namely: cleanliness; elimination of seals; stability or lubricant over wide ranges of temperature; and low friction and heating.

Many systems of classification suggest themselves and the reasons for selecting one system over another are not that overwhelming. However, I thought it might be more interesting if we looked at that overall picture by fields of application. This might have a little more significance.

Medical.

The item of widest use in this area is the high-speed dental drill. More than 20,000 of these have been manufactured. Low friction and low energy losses make these attractive since they operate at about 500,000 RPM (Ref. 29). However, additional advantages over ball bearings are quietness, freedom from vibration and elimination of oil mist in the patient's mouth. The high-pitched noise associated with ball bearings could actually be detrimental to the hearing of those exposed to it for long periods of time. Material studies were needed before these devices were perfected since the load applied to these bearings is entirely up to the dentist's discretion. The rate of cutting may introduce overload reactions on the bearings and grounding out of the journal and bushing, braking the drill to an instantaneous halt from a speed of one half million revolutions per minute. This is rather drastic treatment. Several material combinations will survive this kind of abusive load application. These are:

(1) Silver impregnated carbon operating against a hardened steel rotor.

(2) Silver impregnated carbon operating against an ordinary aluminum rotor.

(3) Tungsten carbide bearings versus tungsten carbide sprayed steel shafts.

(4) Aluminum oxide aginst aluminum oxide.

(5) Shurlube (Teflon-coated bronze) against hardened steel.

Maximum journal diameter is limited to about 3/16 in. With a design journal load of approximately 6 ounces, the unit pressure becomes about 13.5 PSI. The bearings are required to be of the hybrid type combining both the self-acting and externally-pressurized contributions to the load-carrying capacity.

Dynamic instabilities have been encountered and synchronous whirl and self-excited whirl have been problems.

Orthopedic surgeons have gained considerable assistance from high speed, air turbine cutters, similar to the dental drill but larger in size. Air bearings present advantages in the reduction of noise and vibration especially with brain and ear surgery. Apparently the hammer and chisel technique is on its way out (Ref. 30).

In another medical area the benefits of low friction are used to permit the measurement of heart action. A ballistocardiograph table has been developed by Mr. H. Roth of the Astro-Space Laboratories in Huntsville, Alabama. The patient is supported by an

externally-pressurized bearing consisting of two 12" diameter discs. The two discs are sections of a sphere with a radius of 24 feet; thus the table and the patient behave as a frictionless pendulum of long radius. The acceleration of the table is continuously recorded by accelerometers using air-lubricated bearings and when multiplied by the mass provides a highly accurate measurement of the cardiovascular forces. Results obtained by using the table have been significant in studying the effect of exercise on the improvement of the cardiac function and protection against coronary heart disease.

Another example of the benefits to be derived from the low-friction capacity of air bearings is in a glaucoma detector. It appears that glaucoma is an all-to-common disease of the eye characterized by an increase in the internal pressure of the eyeball. This leads to atrophy of the optic nerve and blindness. The build-up in pressure is rather gradual and if detected in time the disease can be cured. Apparently the only way to observe this condition is to press against the eyeball with a force-sensitive or pressure-sensitive instrument and measure this internal pressure. A tiny instrument to do this has been developed by The Franklin Institute, Fig. 23 in which the cylinder is supported in essentially a zero friction condition by externally pressurized air bearings. This permits accurate measurement of eyeball pressure.

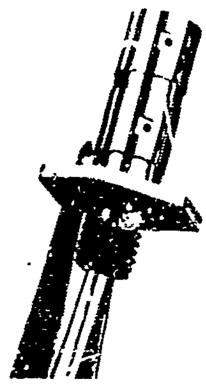


Fig. 23
Stationary Piston for Glaucoma-Detector Device

Dia. - 1/4"; Diametral Clearance - 0.0003";
Length each Bearing - 0.276";
Supply Pressure - 2 psig

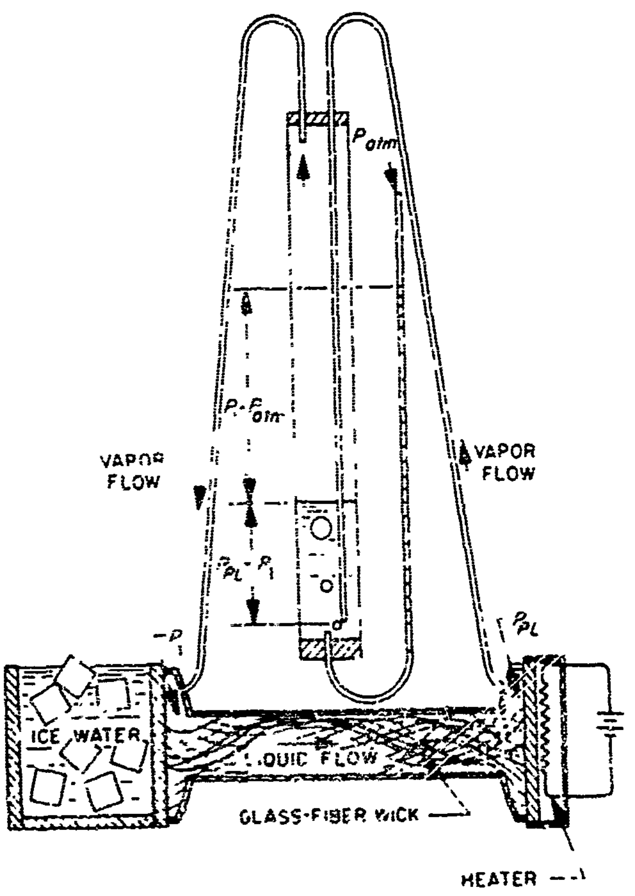
Navigation.

By far the largest application of gas-lubricated bearings is in the inertial guidance field. Practically every manufacturer of gyroscopes is engaged in gas bearing development for his products. One of the most successful of these is Autonetics with a record of having produced many thousands of such devices, but there are many others. The main reasons for interest in gas-lubricated bearings for gyros, are long life, constant friction, high "g" acceleration capabilities, and the possibility of isoelastic design. The bearings that have been developed include a wide variety of shapes and configurations; a late one by Dr. Pan of M.T.I., includes a compliant component. In general the externally pressurized class is used for gimbal mounts and the self-acting variety for the spin axis applications. This is a very important gas-bearing application. Denhard of M.I.T. Instrumentation Laboratory and Pan of M.T.I. list 17 manufacturers of gyros in this country, in England

and France using gas-lubricated bearings and in a paper presented at the 1968 Gas Bearing Symposium list 61 references to published papers on this topic (Ref. 31).

One should also mention the extensive use of gas bearings in accelerometers.

Dr. McGinness of the Jet Propulsion Laboratory has developed a closed-cycle, two phase, thermodynamic system that can use thermal power from solar or nuclear energy sources to evaporate a refrigerant and form a pressurized vapor (Ref. 32). The cycle is completed by condensing the vapor in a condensor and then



Capillary Pump
Fig. 24

pumping it back into the evaporator. Fig. 24. Instead of using a mechanical pump, all moving or rotating elements are eliminated from the system by using a capillary-action pump in the form of a wick. The maximum pressure difference that can be achieved depends upon the effectiveness of the wick. Experimental values as high as 1.52 PSI were reached. A model of an externally-pressurized gas bearing weighing 64 grams was operated continuously and demonstrated the feasibility of such a system. The bearing floated as long as sufficient temperature difference was maintained between the condensor and the evaporator.

The system is of special interest to space navigation because it can operate in a zero gravity environment directly from solar or nuclear heat sources without conversion to electrical energy.

A concept for a new, large-diameter, precision radar antenna is being considered, whose feasibility may largely depend, in turn, upon the practicality of compliant surface bearings. The design is based on the use of a thin walled sphere, 175 feet in diameter. This is the so-called "Eye-Ball" antenna of the Lincoln Laboratories of M.I.T. Very preliminary calculations indicate that the spherical structure might weigh 2.86 million pounds. If a nest of 10 circular hydrostatic pads could be used for the bearing, all located at the bottom, with each pad 24.4 feet in diameter, the projected area of the 10 pads would be $67.37 \times 10^4 \text{in.}^2$. With a 125 MPH

wind adding to the load, the average pressure would be about 7.5 PSI and the maximum pad pressure about 15 PSI. If because of the indeterminant action of the structure, in the worst case only half of the 10 pads carried the load, the average pressure would be about 15 PSI and the maximum about 30 PSI. Apparently the wall stiffness of the light-weight sphere could accept this low pressure without local dimpling.

One could not imagine trying to machine a 24.4 foot diameter bearing pad to conform to the surface of a 175 foot diameter sphere and maintain a film thickness measured in terms of a few mils. This would clearly be impossible even if the surfaces themselves did not deflect elastically under load. However, with a compliant layer on the 10 bearing pads, it appears that a practical solution might be achieved with little concern for tolerances or local deviations from true geometry. The bearing pressures are also low enough to justify the consideration of pressurized air as the lubricating medium.

Manufacturing.

The use of gas-lubricated bearings has caught on well in this area of our industrial activity. Many precision grinding spindles have been built since the first Pratt & Whitney spindles were sold in the early 1950's. These include external and internal grinders. Work heads (TAIL STOCKS), for grinders have also been put on gas

bearings with tremendous reductions in runout and consequent improvement in tolerance control.

Rotary jig and indexing tables are also quite common.

Measuring gauges and jigs are reported in the literature designed to measure roundness of cylindrical specimens, pitch errors of lead screws, etc. The usual stick-slip is completely eliminated. Slideway applications for milling machine tables and grinding machine slides are also included.

One leading manufacturer of gears has designed a testing device supported by externally-pressurized air bearings in which a set of gears is placed, loaded and then run. Because of the gas bearings, the only noise present is that due to tooth contact. This noise can be interpreted in terms of gear quality and precision of manufacture and provides an effective means of inspection. Noisy support bearings would of course, make this kind of determination impossible.

General industrial.

Here there is quite a spectrum of use for gas-lubricated bearings. One could include motor driven compressers, blowers, and fans, turbo compressors and turbine driven pumps and blowers. There are many. Sternlicht in a recent paper (Ref. 33) lists about two dozen of these in this country with a wide range of sizes, speeds and other specifications. Outstanding among them would be

the many blowers and circulators of Societé Rateau in France, listing some 59 machines. They tabulate units running on CO_2, He, Argon and Air, from pressures of 1 to 70 atmospheres, gas temperatures from 10 to 500 C, and speeds from 2,500 to 25,000 RPM. This company recently stated that it had operational experience on its gas bearing equipment of 150,000 hours (Ref. 34). Rotor weights are as high as 550 kg.

In Czechoslovakia there is a 12,000 RPM turbo-compressor rated at 500 kilowatts.

At the opposite end of the size spectrum miniature cryogenic turbo machinery is being designed for gas bearings. Speeds of 240,000 RPM are not uncommon.

The Brayton cycle turbo-compressor of AiResearch running at 50,000 RPM is attracting much interest on its gas bearings.

Whitley lists nine gas-lubricated circulators for gas-cooled reactors (Ref. 35). We should not fail to list air cycle refrigeration machines and air expanders. These have been a very reliable example of the use of gas-lubricated bearings.

Nor should we fail to mention potential uses of these bearings in applications to the chemical and food processing industries. Their freedom from contamination is without peer.

It should be said though that at the present level of gas bearing technology, the application of gas bearings in most cases requires some development work.

Along with the above references in the industrial classification one might mention an in-line flowmeter, placed in a pipe between flanges. Also transport of palletized loads in warehouses and factories, has become an area for the use of gas bearings. An example with amusing possibilities is a domestic refrigerator having a flexible membrane under the base that can be pressurized by a vacuum cleaner with its hose connected to the discharge end. It is called RIDE AIRE. This is almost identical to the commercial Hovair bearing mentioned earlier.

The British report on a similar technique being used under a 220 ton bubble chamber in a high energy laboratory for convenient transportation and location within the laboratory (Ref. 36).

Power generation.

Rotating machinery for turbo alternators, similar to the compressors and circulators mentioned above have found their way into current technology. Again, Sternlicht lists several turbines, turboalternators and motor-generator sets that have been designed and tested and that of course use gas bearings. Auxiliary power turbines are being developed for commercial aircraft use by Allison Division of General Motors. In the power generation category we might also list the gas-lubricated seal development being carried on at Franklin, Pratt & Whitney, and elsewhere for large jet engines. This is an entire field in itself. The application

of gas films to non-contacting face seals can produce leakage rates an order of magnitude lower than that of conventional labyrinth seals used in current rotating machinery. These are especially significant for compressor end and interstages of future large supersonic transport engines. There is continuing activity developing these seals. Presently the hybrid design looks most promising.

Computer industry.

Digital computer hardware has made wide use of the gas lubricated bearing. Probably the first application was for a slider bearing in random access disc files. Rotating drum and disc type memories themselves may be supported by air journal and thrust bearings. And they use magnetic heads operating at fixed distances from the moving surfaces. Tape transport also involves air lubrication as it slides over magnetic heads. This is called a foil bearing, mentioned earlier. Air bearing guides are also used where the tape has to be constrained and turn corners.

Foil bearings can be of the self-acting type, or of the externally pressurized type or the combination of both known as hybrid. Naturally the elasticity of the foil is a significant parameter.

Much recent research work has been aimed at an understanding of foil bearings. As mentioned earlier, Mr. Wildmann of Ampex in a survey paper on foil bearings lists eleven papers, all published since 1965.

Transportation.

One of the current urban problems is to get travelers off the road and into public means of high-speed, safe and reliable transportation. It seems that the dynamics of the wheel and axle and rails imposes some practical limits as to the stable speed that may be attained. This may be in the range of 125 to 150 miles per hour. Above that speed one might suggest looking to alternate means for supporting loads, and introducing guidance and low frictional resistance.

Excluding Hovercraft and air cushion devices, because they are not really gas-lubricated bearings, one thinks of the French development of the train that has been built that rides on a concrete, inverted tee track. This is being developed for the commercial market by the Societé de l'Aerotrain. Some operating experience has been accumulated at this test facility. Very recently a California aerospace company has entered this field with a similar vehicle.

There are naturally a number of problems that will need to be overcome including the smoothness and trueness of the concrete "roadbed," the effect of snow, ice, debris, vandalism, and not the least of all, the presence of birds.

Alternate methods seem much more practical and attractive. Hanging the vehicle from an overhead slotted tube suspended like a monorail for example would permit the load-carrying shoes to be inside of the tube out of the weather and free from vandalism. In

addition the shoes could be made with a compliant layer.

This would reduce the tolerance on surface deviations, and also reduce the pressure and flow requirements for the bearing shoe. At high speeds the bearing might conceivably become self-acting require no external pressurization. This concept shows a lot of promise.

A Look Ahead

What of the future? What do we expect that gas bearing research can provide during the next few years?

We could respond to those questions by compiling a list of technical problems needing attention and areas that need enlightenment, which we will do, but if that were all, it would be taking a rather narrow point of view, and would be ignoring the broader interdisciplinary nature of gas bearing research and its very significant ramifications. It has been especially effective in stimulating interaction of research people in many disciplines for the ultimate benefit of all involved. I think this aspect should be clearly recognized and most definitely emphasized.

In a foreword to the Proceedings of the Second International Symposium on Gas Lubrication. Dr. Beno Sternlicht has made comments that I have extracted and that I would like to quote. He said that very early in our gas bearing research activity,

"A common interest was established in a community of scientists, engineers and educators. Almost from the start, this

community was international in composition. The participants in both the first symposium (1959) and the second symposium (1968) exemplify this international interest.

"U.S. scientists have lectured at several gas lubrication courses in England, and British scientists have participated in courses held here.

"The challenge and interest of the subject brought together diverse talent. Not only specialists in fluid mechanics and mathematics found interesting problems; but physicists, chemists, metallurgists and instrumentation specialists were motivated and made significant contributions (also people interested in design). Individuals with training in over twenty distinct science and engineering disciplines have made contributions in this field. In fact, the major contributions to gas lubrication have required interdisciplinary efforts.

"This is particularly true when equipment employing gas bearings is being developed. In the case of gas bearing gyros, for example, bearing analyses cannot be separated from gyro performance calculations; nor material choice from manufacturing procedure; instrumentation and metrology from gyro testing; or motor and wheel design from friction and windage losses. Similar observations can be made relative to gas bearing turbomachinery, machine tools (orthopedic grinders, the general field of rotor dynamics) and other applications.

"Interdisciplinary effort has also been required in gaining a fundamental understanding of gas lubrication. For example since bearing distortions affect bearing performance, simultaneous solutions of fundamental equations from fluid mechanics, heat transfer and elasticity were necessary. In the case of vapor lubrication heat transfer and thermodynamics relations must also be satisfied. Squeeze film lubrication requires the coupling of compressible flow analysis with resonant vibrations in crystals. In foil and compliant surface bearings, hydrodynamic and elasticity relations must be coupled to take into account the interaction between the fluid film and the bearing surface.

"The overall effort combines basic research, applied research and the development of new or improved applications and products. New businesses based on this technology have been and are emerging in such fields as precision spindles, turbomachinery, medical equipment, gyros and accelerometers, instruments, transportation machines, and metrology equipment.

"In this evolution from research to product, new problems are emerging which provide new challenges and opportunities for further interdisciplinary contributions by the scientific and engineering community. Thus this field of endeavor, will continue to offer research stimulation to some, education to others and exploitation of product opportunities to still others."

End of quote.

I might add, as just one indication of the broad spectrum character of this research activity, and something of its international significance that at the University of Southampton in England, there have been four bi-annual symposia on gas lubricated bearings, each with a published volume of proceedings. This is in addition to the two symposia held in this country. There have also been literally scores of papers published in the technical literature that were presented on other occasions. This is indeed a broad gauge activity that has made itself felt in many disciplines of science and engineering and has much promise to continue to do so.

To make this review as complete as possible I would now like to list some of the specific technical areas in gas-lubricated bearings where further investigation is needed. These are not in any particular order.

1. Explore pneumatic instability of externally-pressurized bearings, both single acting and double acting, with various geometries and source patterns.

2. Develop design coefficients for various externally-pressurized thrust and journal bearings. These would relate load-carrying capacity, film thickness, supply pressure and bearing geometry.

3. Further investigation into multiple source, externally pressurized bearings considering the shock wave phenomenon. Explore the value of elastic orifices.

4. Generalize the mean-free-path effects on thin gas film conditions of lubrication; i.e., gas films of thickness 5 to 100 microinches.

5. Probe into the effects of surface imperfections (micro- and macro-roughness) on steady and dynamic behavior of gas bearings.

6. Investigate surface treatment possibilities for thin film operation where anti-wear coatings may be only 3-5 microinches thick.

7. Obtain a good set of experimental data for crowned, tilting-pad thrust bearings.

8. Establish an understanding of pivot design variations on short and long term behavior of tilting pads for journal bearings and thrust bearings. Include effects of temperature and ambient atmosphere.

9. Explore the influence of external mounting techniques, including stiffness and damping, on the stability of gas-lubricated bearings.

10. Determine design equations for hybrid journal bearings for other than laminar flow conditions.

11. Conduct dynamic whirl studies of journal bearings with various geometrical configurations.

12. Investigate flow analysis for bearings including inertia terms.

13. Analyse vibration response of bearings to external excitation.

14. Investigate on an analytical basis the use of porous bearing materials for externally-pressurized bearings including the damping effect of the pores and the slip effect due to hole size.

15. Study in a more general way the behavior of vapors in gas lubricated bearings.

16. Consider the analysis and development of large diameter, non-contacting seals.

17. Recognize that bearing designs for gyro applications require much further development.

18. Explore the possible value of foil-supported, high-speed rotors.

19. Study the potential use of compliant surface, gas lubricated pads for support of transportation vehicles.

20. Investigate the general facets of behavior of non-rigid (compliant) surface bearings on gas bearing performance.

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This report is the substance of a talk presented before representatives of ONR in Washington, D.C. on January 27, 1970. The review covers results of research in a spectrum of gas-bearing configurations, applications of gas bearings in various use categories, and future prospects.

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